

Relative Change in SI Engine Performance Using Hydrogen and Alcohol as Fuel Supplements to Gasoline

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ABSTRACT

A simulation study on the effect of hydrogen and ethanol addition as supplementary fuel for gasoline engine at lean mixture (equivalence ratio $\phi = 0.8$) was carried out to reduce the gasoline share in the mixture, thus reducing the fuel consumption and harmful emissions. The effect of supplementary fuels on engine performance, emissions, and availability was investigated. This was done by changing the ratio between gasoline and the supplementary fuels in the fuel mixture to achieve the required equivalence ratio. The first part of the simulation consisting of the performance and emissions calculated using the first law, was conducted for all engine speeds. The second part consisting of an availability analysis was conducted at the rated speed of 2750 rpm. The simulation study was conducted using the data obtained from measurements of Ricardo E6/T engine parameters (variable compression ratio engine). The data was also used to verify the models. The study shows that the hydrogen addition reduced the carbon monoxide (CO) and nitrogen oxides (NO₂) share at the lean mixture. The hydrogen addition significantly improved the heat release rate compared with pure gasoline; however, the heat released was close to the top dead center due to its fast burning speed. The ethanol addition improved the first law performance of the engine, e.g., power and efficiency; however, at the cost of increased heat loss. It also improved the indicated work availability in comparison with the addition of hydrogen.

Keywords: hydrogen, fuel blend, ethanol, engine availability, engine performance, emissions, lean mixture, entropy.

INTRODUCTION

The world is consuming its conventional energy sources (mostly fossil) at rapid rates to meet its growing demand. Due to their nature of usage, the consumption of such fossil fuels resulted in environmental problems that have reached an alarming rate. This problem manifested itself in greenhouse gases, depletion of the ozone layer, acid rain, etc. This necessitates the need to find more eco-friendlier energy sources. Several properties possessed by hydrogen made it more preferable to the conventional spark ignition (SI) engine fuels, i.e., gasoline [1–6]. Hydrogen has four times the diffusion coefficient of gasoline; this helps better mix and more homogeneity of the fuel mixture [3, 7]. The adiabatic

flame speed of hydrogen is nearly four times higher than that for gasoline; this means more stable engine operation and combustion process closer to constant volume. The flame development and propagation durations are reduced with hydrogen and increase the engine thermal efficiency [1, 2, 8]. On the other hand, higher adiabatic flame temperature means higher NO₂ levels. Furthermore, hydrogen suffers from another severe drawback, i.e., the fuel energy density calculated on a volume basis is much lower than gasoline. This adversely affects the engine out torque. Higher flammability range and lower minimum ignition energy compared with gasoline make the hydrogen–gasoline blend easier to ignite and improve combustion stability under lean conditions.

Hydrogen as a fuel for internal combustion engines (ICE) has been subjected to significant research. They focused on its use alone and as a supplement to other fuels. Furthermore, some researchers studied the use of hydrogen in spark ignition (SI) and compression ignition (CI) engines [9–13].

An extensive review of hydrogen as a fuel for internal combustion engines is presented in [14–17]. The significant advantages and effects on engine performance as well as the main difficulties and problems in implementing hydrogen were also discussed. The effect of various engine parameters at constant speed (variable load) or constant load (variable speed) conditions were studied in the literature. The studies also include the influence of compression ratio (CR), combustion chamber surface/volume (S/V) ratio, and the boost pressure on engine performance [18]. They reported an increase in engine efficiency and a reduction in NO₂ level with a decrease in the S/V ratio.

Martinez-Boggio et al. [18] reported an optimum hydrogen/gasoline blending percentage with gasoline (by volume) that prevents cycle-to-cycle variations. This percentage is a characteristic of a specific air-fuel ratio. Mohammad Kamil et al. [20] found that the engine output power dropped less than 6% for a 10% hydrogen addition to gasoline. Salahaddin et al. [21] reported that the addition of ethanol to gasoline (20% by volume) did not improve the engine performance. However, the combustion process, its efficiency (defined as the amount of fuel energy supplied to the control volume around the engine which can be released by combustion), and stability (defined as that the variations of the combustion process from cycle to cycle) improved with hydrogen addition to the gasoline-ethanol blend. On the other hand, it caused the NO_x emissions to increase.

D'Andrea et al. [22] studied the influence of hydrogen blended with gasoline at different engine speeds and for the range of equivalence ratios on the combustion process. They reported lower cyclic variations and decreased combustion durations with hydrogen addition. Iodice et al. [23] showed that the addition of ethanol to gasoline resulted in reduced cold emissions. Phuangwongtrakul et al. [24] also showed improvements in engine torque and efficiency with ethanol and gasoline in the SI engine. Varde [25] studied the engine operational

stability using hydrogen-gasoline blends. He found that the combustion durations were reduced with the extension on lean burn limits with hydrogen addition.

Dimopoulos et al. [26] also proved that with the help of hydrogen injection, pilot fuel greenhouse emissions could be reduced. Other researchers [27, 28] showed improvements in engine thermal efficiency and emissions except for the HC and CO emissions at around 5% hydrogen volume fraction. Several studies [29–38] showed a reduction in some regulate emissions, e.g., carbon monoxide (CO), total hydrocarbons (THC), and carbon dioxide (CO₂) with ethanol addition. On the other hand, no significant trends were reported in these studies regarding the NO₂ emissions connected with ethanol addition. The major disadvantage reported as a result of ethanol addition is that it leads to an increase in the formaldehyde and acetaldehyde emissions.

As far as the second law analysis is concerned, Hakan Ozcan [39] conducted a theoretical analysis of hydrogen as supplement fuel for CNG for SI engine fueling. He concluded that the hydrogen addition had a noticeable effect on engine availability (It is defined as the maximum work that can be produced from the system through the interaction with its surroundings during a reversible transition to a state of thermal, mechanical, and chemical equilibrium with its environment, which is defined as the dead state of the system). Combustion irreversibilities decreased while the thermal efficiency, calculated using the second law model, increased with hydrogen addition. Rakopoulos and Kyritsis [40, 41] studied the effect of hydrogen enrichment on the second law performance of an engine powered with natural gas and landfill gas. They also reported a reduction in combustion irreversibility and, as a consequence, an improvement in “second law efficiency” with increasing proportions of hydrogen.

The main objective of this research was to investigate the effect of adding hydrogen and ethanol to gasoline on the SI engine performance from the first and second law point of view. The experiment was conducted for the lean air/fuel mixture ($\phi = 0.8$).

Fuel Properties

Table 1 shows some of the significant properties of hydrogen and ethanol as fuels for SI

Table 1. Properties of fuels [31÷35]

Property	Gasoline	Hydrogen	Ethanol
Density at 1 atm and 300 K (kg/m ³)	720 - 760	0.082	796
Stoichiometric A/F ratio (by volume)	59.5	2.39	14.28
Research octane number	90 - 100	130	108
Lower heating value (MJ/kg)	43.448	119.7	26.952
Thermal conductivity at 300 K (mW/m K)	11.2	182.0	171
Diffusion coefficient into air at NTP (cm ² /s)	0.05	0.61	0.115
Flammability limits (% Volume)	1.2 - 6.0	4 - 75	4.3 - 19
Minimum ignition energy (mJ)	0.25	0.02	0.23
Laminar flame speed at NTP (m/s)	0.37 - 0.43	~2.0	0.62
Adiabatic flame temperature (°C) in air	~2470	2318	2082

engines. For stoichiometric combustion, hydrogen needs (on a volume basis) the least amount of the air (2.39 moles) compared to 14.28 moles for ethanol and 59.6 moles for gasoline. On the other hand, on a mass basis, it needs the highest amount of the air for stoichiometric combustion (34 kg) for hydrogen compared with 8.95 kg for ethanol and 14.7 kg for gasoline). It has the highest calorific value on a mass basis (lowest value on a volume basis). It means the highest energy, which could be released during stoichiometric combustion.

Hydrogen has the highest octane number amongst other fuels, which contributes to its excellent knock resistance. It has excellent diffusive and heat transfer characteristics. Hydrogen has the widest flammability limits, which enables it to burn leaner air/fuel mixtures and hence improves the engine fuel economy. The lower minimum energy value needed for ignition makes it burn faster and better than other fuels. It may be concluded that all three fuels have significantly different combustion properties.

The majority of IC engines are designed and optimized for gasoline (or Iso-Octan) as the prime fuel. The use of any other fuel or fuel blends sometimes demands changing its construction and practically always regulating a fuel system. The engine adaptation allows mitigating some problems, e.g., excessive rate of pressure rise (as a result of speedy combustion process) or premature completion of the combustion process near the top dead center (TDC).

One way of proper utilization of the supplementary fuel is to understand how it affects the engine operation. Most of the above-presented research focuses on the alternatively fueled engine parameters, and the engine cycle is analyzed

using the first thermodynamics law. This study combines both the first and second law analysis to calculate engine parameters for fueling with two alternative fuels as supplement fuel to gasoline at lean operation.

The engine examination aimed to analyze the engine thermal efficiency for gasoline fuelling and compared it to the proposed engine fueling methods with two fuel blends (hydrogen/gasoline and ethanol/gasoline). The equivalence ratio selected for this study was $\phi = 0.8$. This regulation enables us to obtain lower NO₂ and CO₂ emissions than for richer mixtures. The study will be continued for lean mixtures to extend the lean limit of the engine run as well as to reduce fuel consumption and engine emissions.

The Engine

The engine used for this study to verify the mathematical model was the Ricardo E6/T, 4-stroke, water-cooled variable stroke (or compression ratio) engine. The model is well verified for this engine and has been tested for other fuels and variables. The basic engine specifications are shown below in Table 2.

Table 2. Engine specification

Engine parameter	Value
Type	Ricardo E6/T variable compression ratio
Bore × stroke	76.2 × 111.125 mm
Displacement volume	504 cm ³
Engine speed	Variable
Compression ratio	8:1 (fixed)
Spark advance	20° bTDC
Throttle position	Wide open (WOT)

Theoretical analysis

A two-zone internal combustion engine model, mainly described in [6, 26] and their relative modifications, was used in this work. This model treats the combustion chamber to consist of two distinct zones (e.g., burned and unburned) separated by the flame front. The main equations applied to analyze those zones are the first law of thermodynamics and the continuity equation. These equations are derived with respect to the crank angle (θ) to yield several coupled first-order differential equations for pressure, volume, temperature (burned and unburned), mass, heat flux, etc. that are applied for both zones. Derivation of the equation of state with respect to θ gives:

$$\frac{dP}{d\theta} = \frac{\left[-\left(1 + \frac{R}{C_v}\right) \cdot P \cdot \frac{dV}{d\theta} - \frac{R}{C_v} \frac{dQ_{cr}}{d\theta} + \frac{R}{C_v} \frac{dQ_{ht}}{d\theta} \right]}{V} \quad (1)$$

$$\frac{dT}{d\theta} = T \cdot \left(\frac{1}{P} \frac{dP}{d\theta} + \frac{1}{V} \frac{dV}{d\theta} \right) \quad (2)$$

where: P is the cylinder pressure [kPa], T is the cylinder temperature [K], θ is the crank angle [degree], V is the cylinder volume [m^3] which is a function of crank angle rotation, R is the gas constant [kJ/kg-K], C_v is the specific heat at constant volume [kJ/kg-K], Q_{cr} is the total heat flux lost to crevice (or with blow-by), and Q_{ht} is the total heat flux lost to coolant [kW].

As for the cylinder pressure, it is usually assumed to be uniform throughout the cylinder charge. The instantaneous cylinder volume $V(\theta)$ measured from bottom dead center (BDC) position can be expressed using the cylinder mechanism (a slider-crank mechanism) with certain modifications as:

$$V(\theta) = V_s \left[\left(\frac{CR}{CR-1} \right) - \left(\frac{1 - \cos(\theta)}{2} \right) + \left(\frac{CRL}{S} \right) - \frac{1}{2} \sqrt{\left(\frac{2 \cdot CRL}{S} \right)^2 - \sin^2(\theta)} \right] \quad (3)$$

where: CR is the compression ratio [dimensionless], CRL is the connecting rod length [m], S is the stroke length [m], and V_s is the stroke volume [m^3].

Derivation of equation (3) with respect to θ gives the rate of change of cylinder volume with crank angle as follows:

$$\frac{dV}{d\theta} = \frac{1}{2} V_s \sin \theta \left(\frac{\cos \theta}{\sqrt{\left(\frac{2 \cdot CRL}{S} \right)^2 - \sin^2 \theta}} - 1 \right) \quad (4)$$

This helps calculate the rate of change of indicated work W with crank angle:

$$\frac{dW}{d\theta} = P \frac{dV}{d\theta} \quad (5)$$

Moreover, the heat lost during the cycle Q_{ht} is used as follows:

$$\frac{dQ_{ht}}{d\theta} = \frac{h_w}{\omega} A_w(\theta) [(T - T_w)] \quad (6)$$

where: A_w is the cylinder wall surface area [m^2], T_w is the cylinder wall temperature [K] and ω is the angular velocity [s^{-1}], and h_w is the convective heat transfer coefficient [W/m^2-K].

The formula used for the calculation of the heat transfer coefficient h_w from cylinder contents to walls is shown below in equation (7).

$$h_w = \frac{P^{0.8} v^{0.8}}{T^{0.53} D^{0.2}} \quad (7)$$

where: the value of the gas velocity v is taken to be equal to $6.18 \cdot v_m$ during scavenging and intake stroke (m/s); and during the compression stroke, it takes the value equal to $2.28 \cdot v_m$. During the remaining four strokes, the value used for the gas velocity is $v = 2.28 v_m + 0.00324 (V_s T_a) / (P_a V_a) \cdot (P - P_{motored})$.

The mean gas velocity v_m is given by $v_m = 2S \cdot N/60 \cdot P_a, T_a, V_a$ are pressure, temperature, and volume of the cylinder at the beginning of compression, respectively; and, $P_{motored}$ is the cylinder pressure during motoring (no combustion) (Pa). As for the energy flow through the crevice, Gatowski et al. (1984) expression for a spark-ignition engine was used. This is shown below in equation (8).

$$\frac{dQ_{cr}}{d\theta} = (u + R \cdot T) \cdot \frac{dm_{cr}}{d\theta} \quad (8)$$

where: u is the internal energy in (kJ/kg), m_{cr} is the mass flow through the crevice in (kg), $dm_{cr} > 0$ for the case of flow into the crevice, $dm_{cr} < 0$ for the case of flow out-of-crevice, $(u + R \cdot T)$ is calculated based on engine cylinder conditions when $dm_{cr} > 0$ and under crevice conditions when $dm_{cr} < 0$.

Further, Wiebe function Eq. (1) [41] was used to model the mass fraction burned (χ_b) vs. crank angle (CA).

$$\chi_b = \frac{m_b}{m_{mix}} = 1 - e \left[-a \left(\frac{\theta - \theta_o}{\Delta\theta_b} \right)^{m+1} \right] \quad (9)$$

where: m_b is the mass fraction burned and m_{mix} is the total mass of the fuel inside the cylinder [kg], θ_o is the start of combustion [deg], $\Delta\theta_b$ is the burning duration [deg], and a and m are model constants.

The derivative of the mass fraction burned as a function of crank angle is given by:

$$\frac{dx_b}{d\theta} = \left(a \frac{m+1}{\Delta\theta_b} \right) \left(\frac{\theta - \theta_o}{\Delta\theta_b} \right)^m e \left[-a \left(\frac{\theta - \theta_o}{\Delta\theta_b} \right)^{m+1} \right] \quad (10)$$

The rate of work availability A_w is given by:

$$\frac{dA_w}{d\theta} = (P - P_o) \frac{dV}{d\theta} \quad (11)$$

where: P_o is the dead state pressure (usually the atmospheric pressure [kPa]).

The expression for the rate of availability transfer due to irreversible heat transfer A_L is given by:

$$\frac{dA_L}{d\theta} = \left(1 - \frac{T_o}{T} \right) \frac{dQ_{ht}}{d\theta} \quad (12)$$

where: T_o is the dead state temperature (or the atmospheric temperature) [K].

The rate of fuel availability A_f into the process:

$$\frac{dA_f}{d\theta} = (1 - x) \left(\frac{dx_b}{d\theta} \right) a_f \quad (13)$$

$$a_f = LHV_f \left[1.0334 + 0.0183 \frac{H}{C} - 0.0694 \frac{1}{C} \right] \quad (13.1)$$

For the closed part of the cycle in an engine, the exergy efficiency is shown as follows:

$$\eta_{exergy} = \frac{W_{net}}{Ex_{fuel}} \quad (14)$$

A detailed presentation of the second law analysis of engine combustion was provided in [43÷47]. In this study, the mixture was varied, keeping the overall trapped equivalence ratio = 0.8. This was done for both fuels. The ignition

timing was kept constant at 20 degrees bTDC as in the engine design. Since the main aim of the research was to keep the fossil fuel used (i.e., gasoline) on a minimum level, the gasoline flow rate was adjusted to keep a constant equivalence ratio ($\phi = 0.8$). The first engine simulation was run for $\phi = 0.8$ with the use of gasoline only. The first calculation enabled to set gasoline dose and a reference engine performance. The gasoline-hydrogen mixture was then created by adjusting (reducing) gasoline mass flow to remain equivalence ratio ($\phi = 0.8$).

RESULTS

The simulation results show the engine performance and emissions, but the result discussion also includes the exergy analysis of the SI engine to maximize the work availability.

Brake Power

Figure 1 shows engine brake power (BP) for ethanol/gasoline and hydrogen/gasoline blends ($\phi = 0.8$). It can be seen from Figure 1 that BP increases with the addition of both hydrogen and ethanol fuels. The influence of ethanol on brake power is more visible than in the case of hydrogen. Several researchers reported an increase of the engine brake power with the addition of ethanol, e.g. [48÷50].

In order to understand this trend, Figures 2, 3, 4, and 5 show the energy flow into the cylinder for all fuel blends, the indicator diagrams calculated for the engine rated power speed,

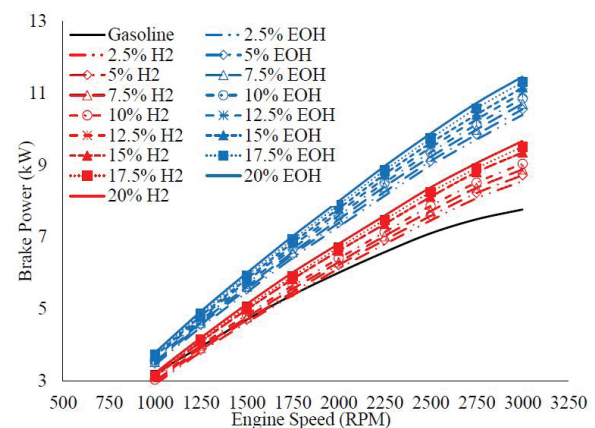


Fig. 1. Brake power variation with engine speed for all fuel blends

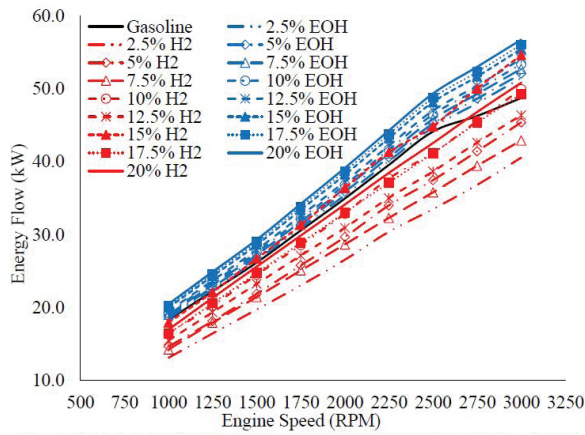


Fig. 2. Energy flow into the engine variation with engine speed for all fuel blends

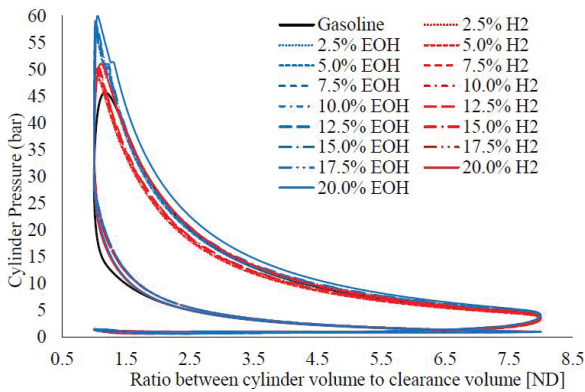


Fig. 3. Indicator diagrams comparison at 2750 RPM

i.e., 2750 rpm and ignition timing 20-degree bTDC) and peak cylinder pressure for both fuels. It can be seen that the energy flow (shown in Figure 2) for both fuel blends increases with the percentage of supplement fuel in the mixture. Furthermore, the amount of energy in the mixture is higher for ethanol/gasoline blend compared with the hydrogen/gasoline blends. In addition, the peak cylinder pressure for both fuels increases with their respective percentage in the mixture. However, this peak pressure occurs close to TDC with higher pressures for ethanol than hydrogen, as shown in Figures 3, 4 and 5. In his review article, Mohammad et al. [52] explained this pressure rise due to the higher volumetric efficiency and lower intake temperatures with ethanol/gasoline blend. There are many reasons for this phenomenon, starting from the mixture energy content flowing into the cylinder, hydrogen properties, and the effect of constant spark timing [1, 2].

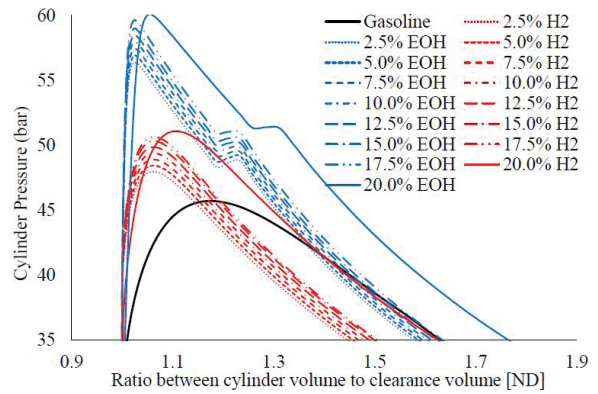


Fig. 4. Peak pressure comparison for all fuel blends at 2750 RPM

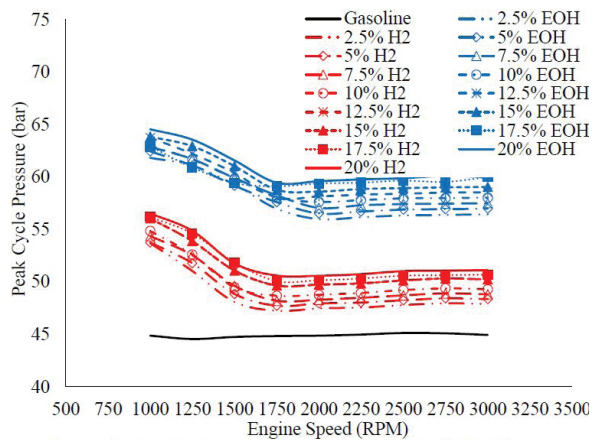


Fig. 5. Peak cylinder pressure vs. engine speed for all fuel blends

Hydrogen tends to decrease the energy content of a cylinder due to lower energy content on a volume basis. In addition, the drop in volumetric efficiency is caused by the heating effect of the inlet manifold, which causes the gaseous fuel to expand. This effect is the opposite of ethanol, which tends to cool the inlet manifold due to its evaporation.

Brake Thermal Efficiency

Thermal efficiency is an important parameter showing how effectively the engine utilizes the energy supplied to it with the fuel to useful work. It can be said that Thermal efficiency is inversely proportional to the engine fuel economy (specific fuel consumption) for the same fuel. As the thermal efficiency of the engine increases, the engine could develop more power for the same amount of fuel. Hence, this factor should be maximized. The relative change in brake thermal efficiency

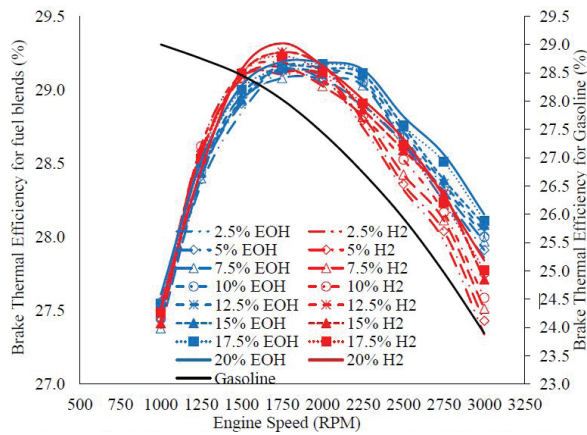


Fig. 6. Brake thermal efficiency vs. engine speed for all fuel blends

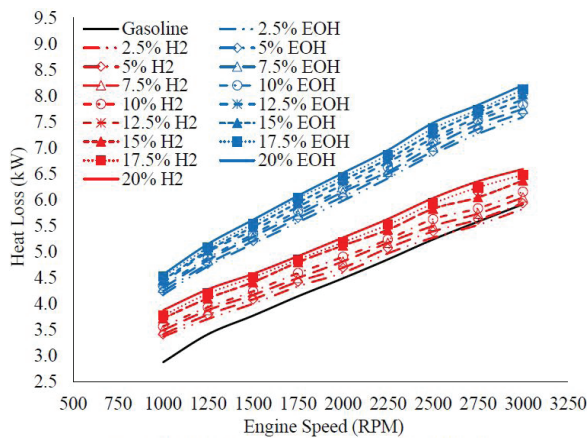


Fig. 7. Heat loss vs. engine speed for all fuel blends

with an engine speed for both fuels is shown in Figure 6. As shown in Figure 6, the brake thermal efficiencies of the hydrogen-enriched mixtures are higher than those with ethanol addition, especially at higher fuel fraction. Moreover, at lower engine speeds, the engine fueled thermal efficiency with both fuel blends is less than that for gasoline.

Deng et al. [51] explained that this is because of the hydroxyl radical (-OH) produced from ethanol, which improves the combustion and flame propagation speed. However, due to the increased amount of fuel in the engine due to the lower calorific value, the energy input rises, and the consequent heat loss increases. Figure 6 shows the percentage of heat loss for ethanol/gasoline blend compared with hydrogen addition to gasoline. For both fuels, the heat loss increases for the mixtures with additional fuels. Furthermore, the heat loss with ethanol/gasoline blends is higher than those

for hydrogen/gasoline mixtures. One possible reason for this phenomenon is the shorter duration of combustion with hydrogen addition due to faster flame speed; hence, less time will be available for the combustion products to lose their energy.

Heat Release Rate (HRR)

Several factors influence the thermal efficiency of an engine: heat loss, combustion time, combustion process (e.g., flame development, duration, and propagation). Figure 8 shows the HRR for all analyzed cases at all engine speeds. As shown in the figure, the heat release rate for both blends is more significant than that for pure gasoline. The figure also shows that the RoHR for the ethanol/gasoline blend is higher than that with hydrogen. The difference between the maximum peak of RoHR for both blends is about 20%, favoring the ethanol/gasoline blend. It was expected due to the higher energy content for the ethanol/gasoline mixture than for hydrogen/gasoline. Further noticed that the angular position of the peak of RoHR shifts more with ethanol added to the mixture and that the effect of ethanol addition is more significant for the case of hydrogen. This can be attributed to the formation of OH radicals in the cylinder, which improves the chain branch reactions and reduces the flame development period [1, 2].

Engine emissions

From the engine emissions point of view, the addition of H₂ helped to reduce the CO emission

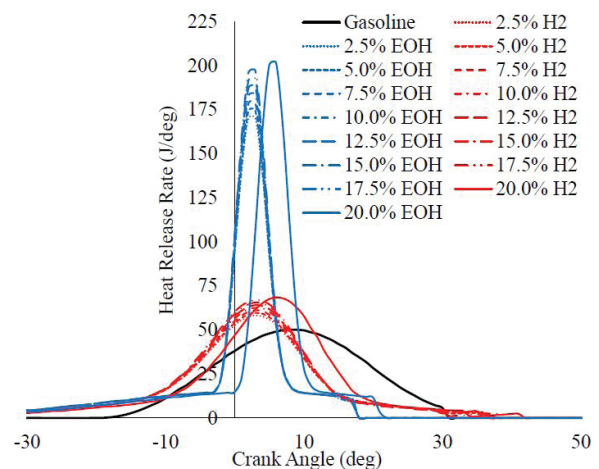


Fig. 8. The comparison of the heat release rate for all fuel blends

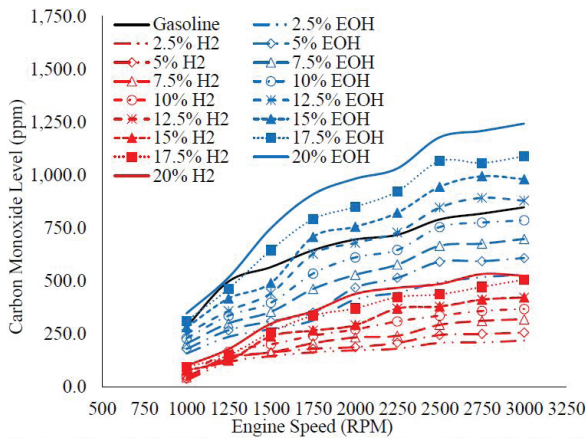


Fig. 9. Carbon monoxide level variation with engine speed for all fuel blends

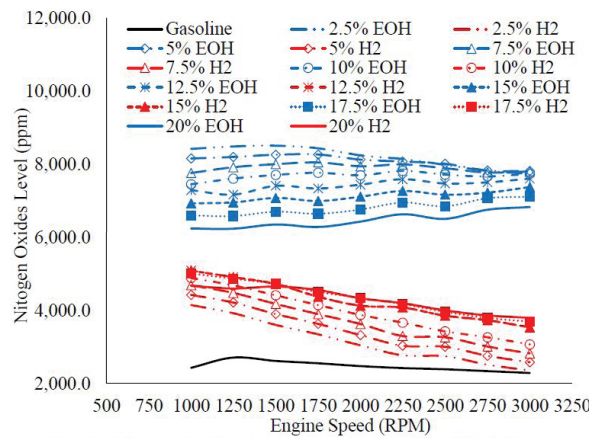
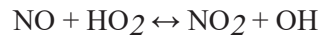


Fig. 10. Nitrogen oxides level vs. engine speed for all fuel blends

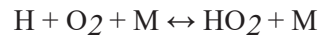
to a greater extent than ethanol. This is shown below in Figure 9. One key factor affecting the reduction of CO formation is the availability of oxygen. Hydrogen enlarges this phenomenon more than ethanol. By adding more hydrogen fuel to the mixture (and reducing gasoline fraction to maintain the equivalence ratio), more oxygen is available, and less carbon is burnt inside the cylinder compared with the addition of ethanol, reducing the carbon-related emissions. Two more factors play an essential role in the increased CO emissions reduction with hydrogen, compared with ethanol. First; the reduction of the overall H/C as explained above, and second is the improvement in the combustion efficiency with hydrogen addition. Concerning Table 1, it can be noticed that the flame speed and the diffusion coefficient for hydrogen are higher than for both other fuels (gasoline and ethanol). These factors help to improve the oxidation process of carbon. Similar results were also reported by [46, 47].

Several factors affect the formation of NO_x emissions inside the cylinder, e.g., combustion temperature, oxygen availability, combustion time, etc. It is noticed from Figures 2 and 3 that the cylinder temperature (and indirectly pressure) increases with the addition of the hydrogen and ethanol to the air-fuel mixture. This ultimately resulted in an increased share of NO_x in exhaust gases, as shown in Figure 10. From the kinetics point of view, NO_x is primarily formed from NO radical and less from NO₂. Nitrogen oxide exists mainly in the flame zone during combustion, and it is then converted to NO₂ as shown in the following reaction:



However, due to the negative activation energy and low levels of HO₂, the NO₂ formation is prompt.

With hydrogen addition, the formation of NO tends to reduce due to the third body reaction shown below:



This causes an increase in the concentration of HO₂ radicals [4]. It is also seen from Figure 10 that the maximum NO_x emissions are reached for 17.5% hydrogen and 2.5% ethanol. Beyond this value, the increment rate is slower. With the increased fraction of the supplement fuel in the mixture, hydrogen needs more air to be burnt. The higher cylinder pressure and a higher temperature are observed in a cycle. One reason for this phenomenon is the phase difference between gasoline and hydrogen, which causes a certain

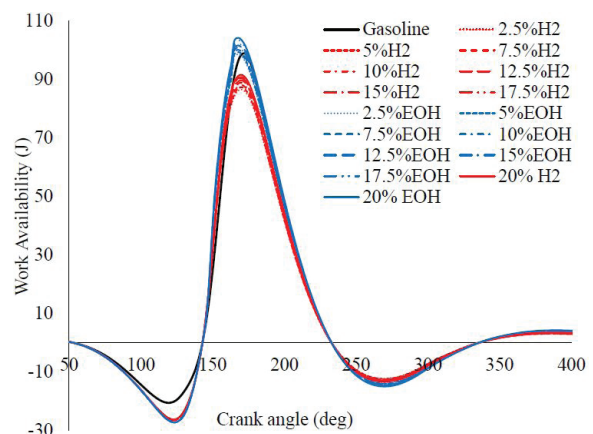


Fig. 11. Work availability

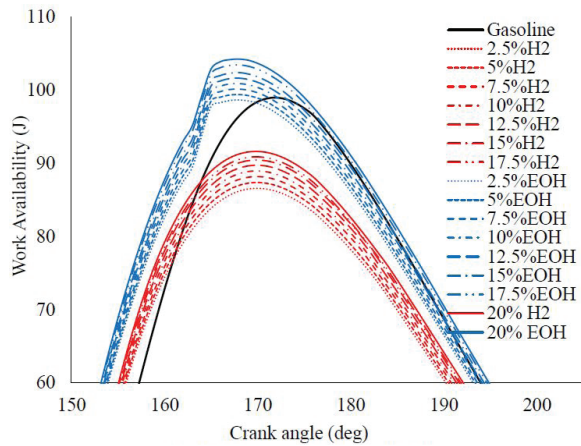


Fig. 12. Maximum work availability

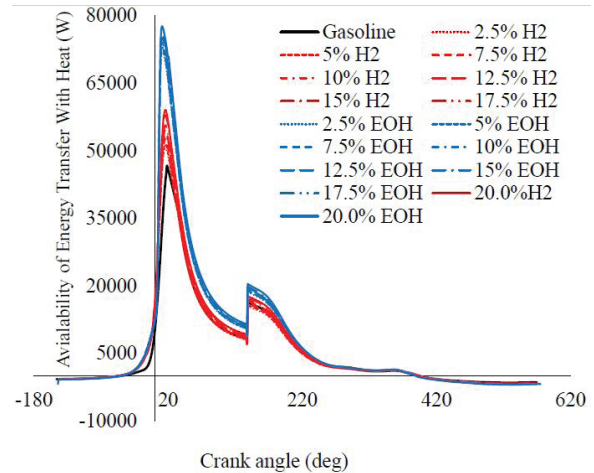


Fig. 13. Availability transfer with heat

degree of the mixture inhomogeneity. It requires supplying a higher amount of air to help to burn these extra hydrogen supplements.

Engine Availability Analysis

The second law analysis shows that the mixture exergy is larger for the ethanol fuel addition than for hydrogen (Figures 11 and 12). Furthermore, Figure 11 shows that for both fuels, the exergy transferred with work inside the cylinder is higher than that for pure gasoline. This trend helps explain why the engine tends to develop more power when fueling with ethanol/gasoline blends at the fixed equivalence ratio for fixed ignition timing than with hydrogen. In addition, Figure 12 shows that the amount of exergy availability inside the cylinder is more with ethanol/gasoline blend compared with hydrogen/gasoline. This helps explain the

above-mentioned first law performance (i.e., engine power and efficiency).

The amount of exergy transferred with heat is shown above in Figures 13 and 14. There is a clear trend that for both additional fuels, the amount of exergy transferred with heat is more significant than gasoline. Further, the exergy transferred with heat for ethanol is slightly higher (as shown in Figure 14) than that for hydrogen. For both additional fuels, the amount of exergy transferred increases the increase in their shares. The results presented in Figure 14 help explain the increase in heat loss for ethanol compared with hydrogen.

Figure 15 shows the cylinder temperature vs. entropy inside the cylinder. This figure helps to understand the heat transfer inside the engine for the engine cycle. For the same temperature, the amount of entropy for ethanol/gasoline blends is lower than that for the H₂ addition. This helps to explain the trend shown in Figures 5 for both

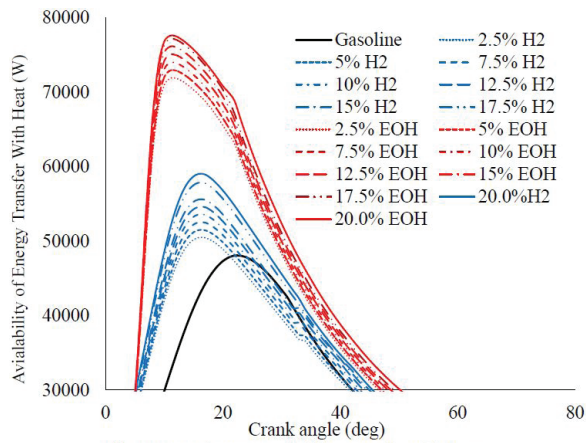


Fig. 14. Maximum availability transfer with heat

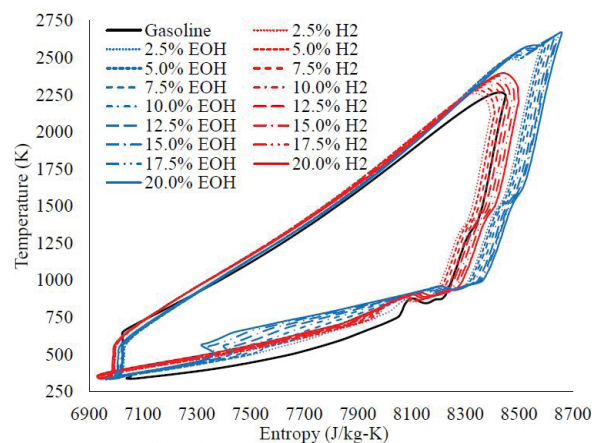


Fig. 15. Temperature-entropy diagram

examined blends where the thermal energy losses are higher, and as a result, engine thermal efficiency drops as a result of higher heat transfer. On the other hand, the area under the curve (Figure 15) for the ethanol/gasoline blends is more significant than that for the H₂ addition. This explains the trend shown in Figures 1 and 7 since the area can be understood as the total work output of the engine. Similar results are shown by [6].

CONCLUSIONS

A study on the effect of hydrogen and ethanol addition as supplementary fuel to gasoline at lean mixture ($\phi = 0.8$) was conducted. The engine performance, as well as engine emission, were calculated with the use of the first law of thermodynamics model. Further, some results of a calculation using the model based on the second law were presented.

The study clearly shows that for lean mixtures, the hydrogen addition reduced the CO and NO_x emission levels, compared with the ethanol addition. The hydrogen addition significantly improves the heat release rate (HRR); however, due to its fast burning nature, the heat released was close to the top dead center compared with ethanol.

The results show that the fueling of an SI engine with ethanol/gasoline and hydrogen/gasoline blends gives some benefits. The ethanol addition improves power, engine thermal efficiency, and engine availability, but hydrogen improved the engine emissions.

The research shows that 15% of hydrogen and the same ethanol share in the blends were found suitable for improving the engine performance and emissions. The influence of hydrogen in a fuel blend on the combustion process is more significant than ethanol. The research shows that for engine fueling with fuel blends, it is necessary to adjust the engine regulation parameters, especially ignition timing. It could contribute to higher engine performance and its better emissions.

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